

Effect of suspension system in the lateral stability of railway freight trucks

H Molatefi^{1*}, M Hecht², and M H Kadivar³

¹Railway Engineering School, Iran University of Science and Technology, Narmak, Tehran, Iran

²Department of Rail Vehicles, TU Berlin, Germany

³Department of Mechanical Engineering, Shiraz University, Shiraz, Iran

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Abstract: In the current research an empty freight wagon with BA652 and an old BA661 bogies have been modelled. Non-linear creep forces with spin moment between wheel and rail have been used and also all impacts and friction forces have been modelled. The characteristics of primary suspension systems for BA652 and an old BA661 have been evaluated by means of test. Non-linear equations of motions and kinematical constraints have been solved in time domain and limit cycles have been showed. Both primary and secondary hunting can be seen in responses of wagon. For BA652 there are non-linear and linear critical speeds and also saddle nodes but for old BA661 the situation is too critical and the system can oscillate in very low speeds.

Keywords: freight wagon, BA661, BA652, non-linear, hunting, limit cycle

1 INTRODUCTION

Dynamics of rail vehicles are non-linear [1] and have been considered for several decades. In freight trucks dry friction, impacts and also the contact between wheel and rail generate non-linear kinematical and dynamical terms [2–7]. Usual freight bogies have two wheelsets, which are parallel and are connected to the bogie frame. The wagon weight is transferred to the track through vehicle's suspension system that has essential role in the behaviour of wagon. Hunting stability is primarily a function of the bending and shear stiffness between two wheelsets. Bogies with low bending stiffness generally are known as 'steering bogies' [8]. These bogies use the longitudinal creep forces to deflect the longitudinal springs, which create the bending stiffness. This permits the wheelsets to align with an almost radial position on the curves. BA type bogies are known as steering axle bogies and Y25 type bogies are known as rigid axle bogies [9]. Critical speed and limit cycles in the dynamics of an empty Y25-freight truck have been analysed in reference [10]

by authors. In the current paper lateral stability of an empty freight wagon with BA652 and an old BA661 bogies have been analysed.

Leaf or parabolic springs have been used from 1890 and have been adapted in bogie suspension systems since 1925. Dry friction between leaves generates damping in these springs. BA652 and BA661 have been designed for axle load of 22.5 ton (Fig. 1). Chain links (or swing hangers) connect the bogie to the leaf spring of the suspension system that rests on the axle box (Fig. 2). This arrangement allows the axle box to move in both longitudinal and lateral directions. As Fig. 2 shows in addition to friction between leaves of parabolic springs there are frictions between bolt and parabolic spring, bolt and chain stone and also chain link and chain stone. Motions are limited only by the play of the axle box in the axle horns that is ± 6 mm in longitudinal and ± 23 mm in lateral directions. Main differences of primary suspension systems of BA661 and BA652 are in their leaf springs and length of chain links (Fig. 2).

Empty wagons are one of the sources for noise in the freight transportation. They have low critical speed and can oscillate with high amplitudes. In this research completely non-linear equations of motions and constraints for BA661 and BA652 freight trucks with empty wagon and 25° of freedom have been derived.

*Corresponding author: Railway Engineering School, Iran University of Science and Technology, PO Box 14515/1179, Narmak, Tehran, Iran. email: molatefi@iust.ac.ir or molatefi@gmail.com

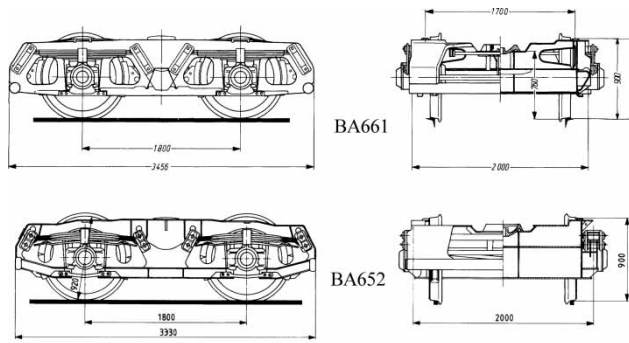


Fig. 1 BA652 and BA661 freight bogies

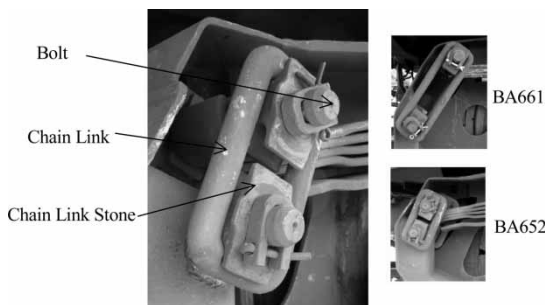


Fig. 2 Chain links in BA661 and BA652

The property of primary suspensions had been tested in TU-Berlin, all dry frictions have been modelled by Kolsch-method and impacts in clearances have been modelled by non-linear springs. Table 1 shows values of masses, position for centre of masses, and moment of inertias for empty wagon of Eaos807.

Wheel/rail contact creep coefficients and kinematical constraints have been derived by means of RSGEO software. Wheelset and rail have been defined in Table 2.

2 SIMULATION OF FORCES

2.1 Forces in primary suspension

Impacts between parts after clearances have been modelled by non-linear springs with high stiffness of

Table 2 Wheel and rail definitions

Wheel profile	Wheel base	Wheelset tape circle distance	Wheel radius
S1002	1800 mm	1500 mm	460 mm
Rail profile	Rail gauge	Inclination	Type
UIC60	1435 mm	1/40	Straight/no irregularities

10^7 N/m (Fig. 3). The stiffness must be high enough to simulate solid contact.

A measurement structure (Fig. 4) for primary suspension systems with parabolic springs has been developed in TU-Berlin [11]. Results show that there are hysteresis loops in three directions. Hysteresis can be assumed as in Fig. 5 for different vertical preloads. Table 3 shows the results for vertical load of 18.9 kN.

To model primary suspension force elements, Kolsch-method [12] has been used that explained in reference [10].

2.2 Forces between carbody and bogies

Bogie frame is connected to the carbody through central pivot. Figure 6 shows the central pivot as a

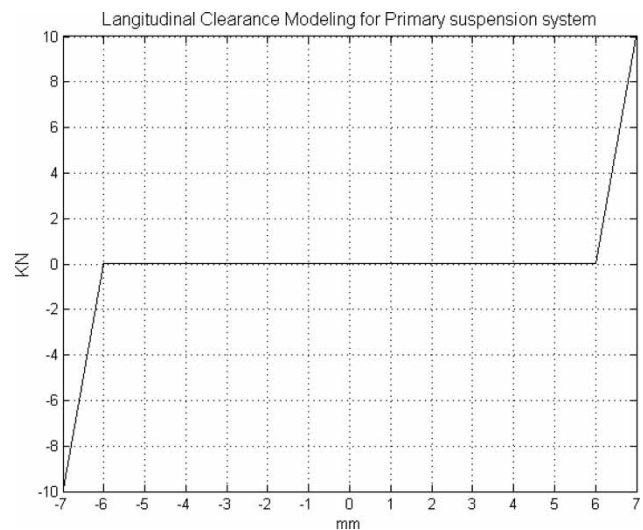


Fig. 3 Non-linear spring for axle box clearance in longitudinal direction

Table 1 Mass, moment of inertia, and centre of mass for the freight truck

	Center of mass ^a (m)	Moment of inertia (kg m ²)			Mass (kg)	
		I_{xx}	I_{yy}	I_{zz}	One unit	
Wheelset with axle box	$h_{wh} = 0.46$	825	121	837	1300	
Bogie	$h_b = 0.66$	1372	1589	2875	2150	Bogie: 4750
Carbody	$h_c = 1.31$	11 389	219 667	225 378	11 400	
Empty wagon	0.9648				20 900	

^aCentre of mass is with respect to the railhead surface

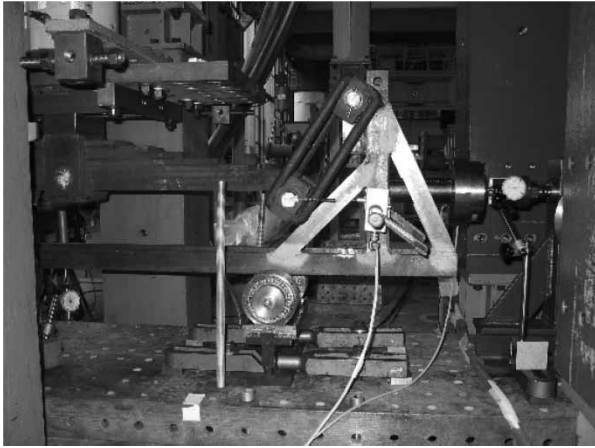


Fig. 4 Test structure for primary suspensions with leaf springs

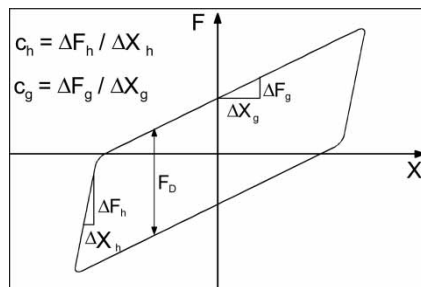


Fig. 5 Hysteresis in the force-displacement diagram

spherical joint. The bogie frame has angular motions with respect to carbody about the joint centre. Contact surfaces and their relative motions generate dry friction in the central pivot. Researches and tests show that the moment forces in the central pivot have maximum value [11] that can be evaluated as equation (1). These values are in the slip mode and in the stick mode the angular stiffness is about 10^7 Nm/rad (0.22 [11] has been used for dry friction coefficient).

$$\begin{aligned} M_x &= M_y = 0.17 \cdot \mu_{\text{CentPiv}} \cdot F_N \\ M_z &= 0.18 \cdot \mu_{\text{CentPiv}} \cdot F_N \end{aligned} \quad (1)$$

Figure 7 shows side bearers; 3 mm clearance has been considered in simulations, which is approximately equal to 0.0035 rad relative roll-movement of

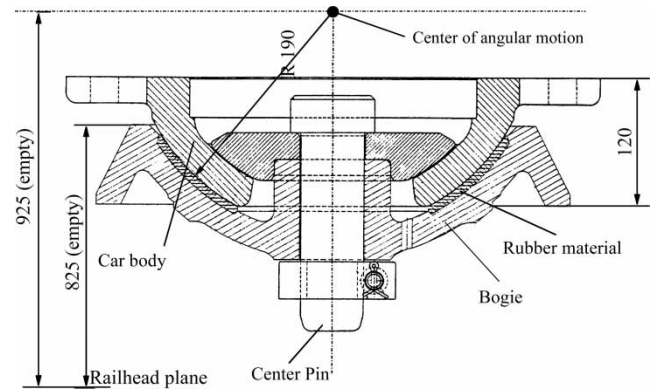


Fig. 6 Central pivot (all dimensions in mm)

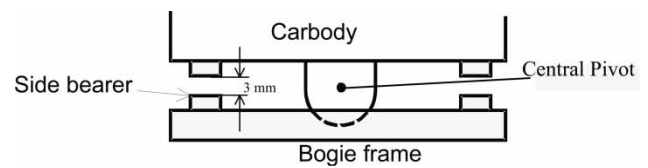


Fig. 7 Side bearer

carbody and bogie. Dry frictions between side bearers and carbody have been modelled using Kolsch-method with 0.22 for dry friction coefficient and 10^6 N/m for stick stiffness.

2.3 Wheel/rail contact forces

Non-linear creep forces model including the effect of spin moment [13, 14] have been used to model wheel/rail contact forces. This model has been explained in reference [10].

3 EQUATIONS OF MOTIONS

As common, rotations around longitudinal, lateral and vertical axis have been defined as roll (ϕ), pitch (χ), and yaw (ψ). Wheelsets have four degrees of freedom (longitudinal, lateral, pitch, and yaw motions). Vertical and roll motions of wheelsets are dependent to lateral and yaw motions of wheelsets and can be calculated from kinematical constraints between wheelset and track. Bogies have four degrees of freedom (lateral, vertical, roll, and yaw motions). Track is straight without irregularity and the pitch motion of bogie frame can

Table 3 Test results for BA652 and BA661 suspensions in empty wagon (Fig. 5)

	X			Y			Z		
	C_h (N/m)	C_g (N/m)	F_D (kN)	C_h (N/m)	C_g (N/m)	F_D (kN)	C_h (N/m)	C_g (N/m)	F_D (kN)
BA661	6.7×10^5	2.2×10^5	1.6	1.5×10^5	1.0×10^5	0.6	7.2×10^6	2.3×10^6	13.0
BA652	9.7×10^6	4.3×10^5	3.2	8.6×10^5	2.1×10^5	1.2	4.3×10^6	9.0×10^5	7.0

be neglected. Carbody (with bogies) is massive and so in the longitudinal direction acceleration is ignorable and so longitudinal velocity is considered to be constant or known for bogies and car body. Assuming the spherical joint in central pivots of carbody reduces the freedom of carbody. Carbody has one degree of freedom (roll motion) around connecting line of its central pivots.

In the current research stiffness and damping of primary suspension have been evaluated by means of test and so equations of motions for this freight truck is the same as in reference [10]. Differences are in bogie weight characteristics, side bearers, clearances, and property of primary suspension. As Fig. 7 shows, side bearer of this type of bogies does not have spring, and relative roll motion of carbody and bogie differs with Y25 bogies. Also for relative yaw motion of carbody and bogies, there is no contact surface with dry friction in the side bearer especially for the straight track.

In results, numbers 1, 2, 3, and 4, have been used for first, second, third, and fourth wheelsets, 5 and 6 have been used for first and second bogie frame and 7 has been used for carbody.

4 RESULTS

Normally critical speed means non-linear critical speed in the railway vehicle system dynamics. The non-linear critical speed is the lowest speed for which a periodical motion exists and it will be determined by the bifurcation analysis. Periodical motions appear with two saddle node bifurcations. The velocity of the first saddle node bifurcation is the non-linear critical velocity. This velocity can be found by driving with perturbed vehicle at a velocity enough more than critical speed. After some time the solution will be converged to a limit cycle oscillation. The next step is an adiabatic reduction of the velocity that can be done by making the velocity a linear function of time. This reduction is done until passing the saddle node, which is clearly seen as the amplitude of the oscillation suddenly drops to zero.

The dynamical systems of freight trucks are very complicated due to the characteristics of the non-smoothness and the discontinuities caused by impacts and stick–slip motions originating from clearance and dry friction. The time for computation of the necessary

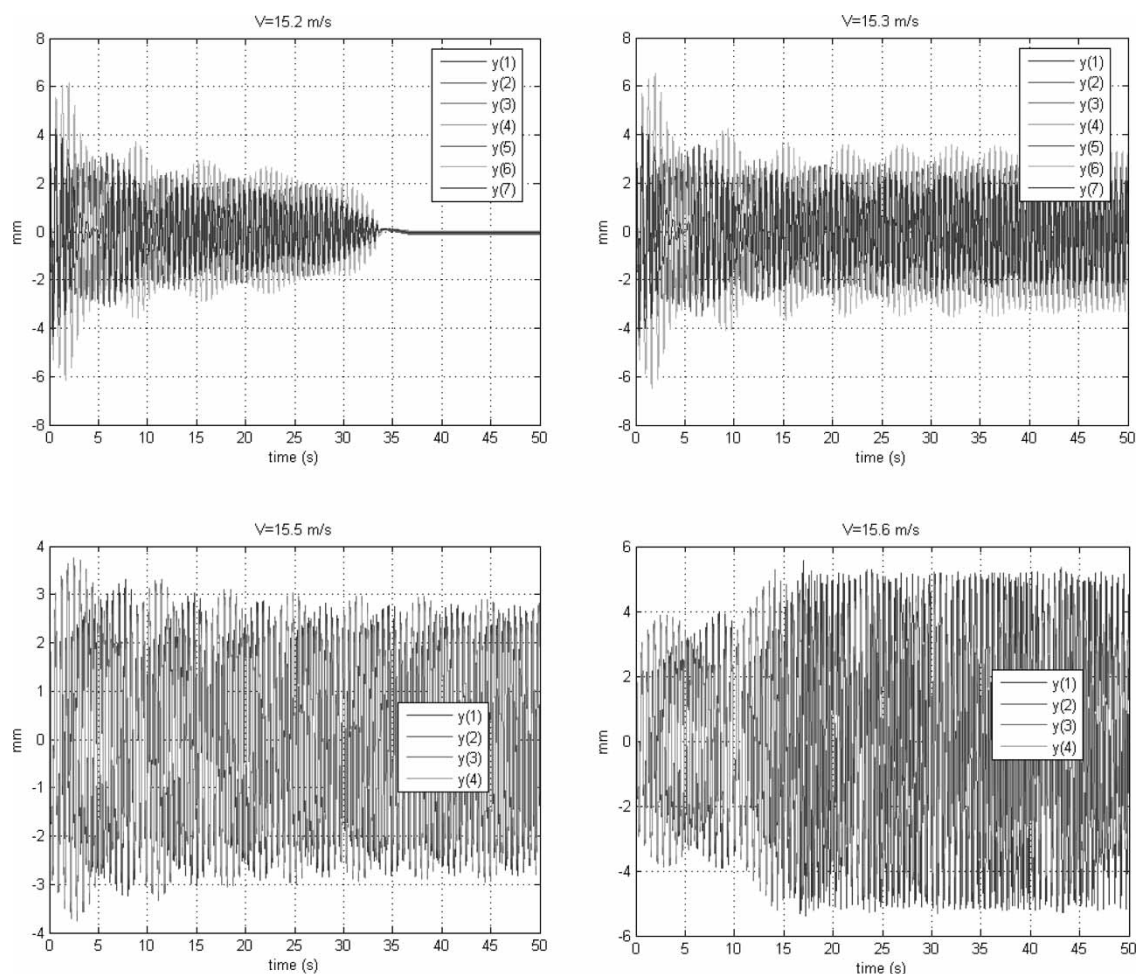


Fig. 8 Search for limit cycles in different speeds

data for fine visualization of bifurcation is very long. To find limit cycles for non-linear systems there is not a unique method. In the current research perturbed wagon has been simulated with different speeds (starting from 1.0 m/s and with step size of 1.0 m/s). With this method limit cycles can be seen and saddle nodes can be found as a jump in the limit cycle amplitude. To find the other saddle node, vehicle has been simulated at speed more than the speed with jumped amplitudes. After 10 s, the speed has been decreased linearly until the amplitudes of oscillations have become zero.

In addition, the speed has been increased to find the behaviour in the speeds more than critical speed.

4.1 Critical speed and limit cycles for BA652

Figure 8 shows the amplitude for lateral oscillations in different speeds and as can be seen the limit cycle appears in 15.3 m/s. There is a jump in the amplitude at speed 15.6 m/s. Amplitudes of lateral oscillations for wheelsets in speeds between 15.6 and 19 m/s are about 5 mm. For 20 m/s these amplitudes are less than 3 mm.

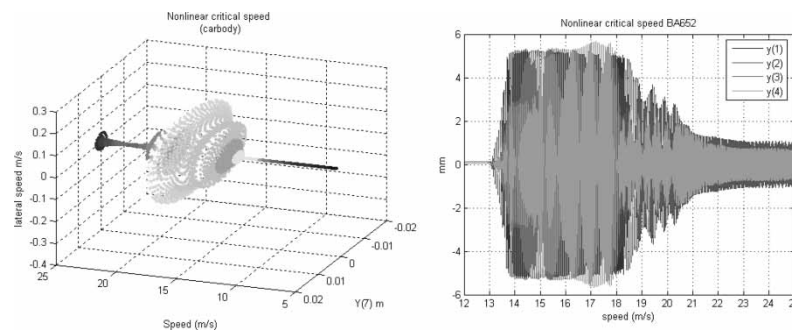


Fig. 9 Non-linear critical speed for the wagon with BA652 bogie

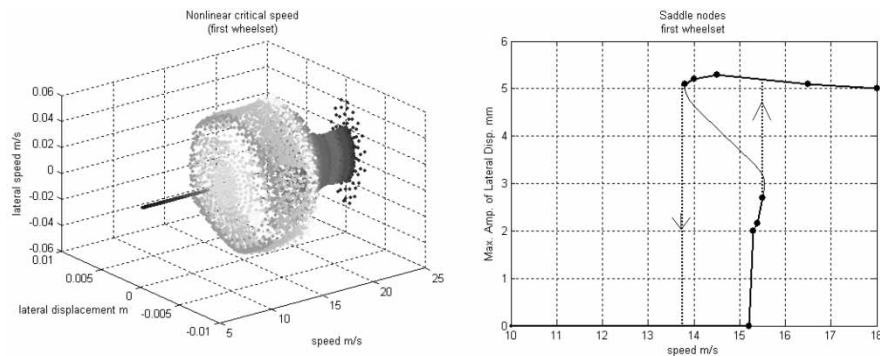


Fig. 10 Non-linear critical speed and saddle nodes (BA652)

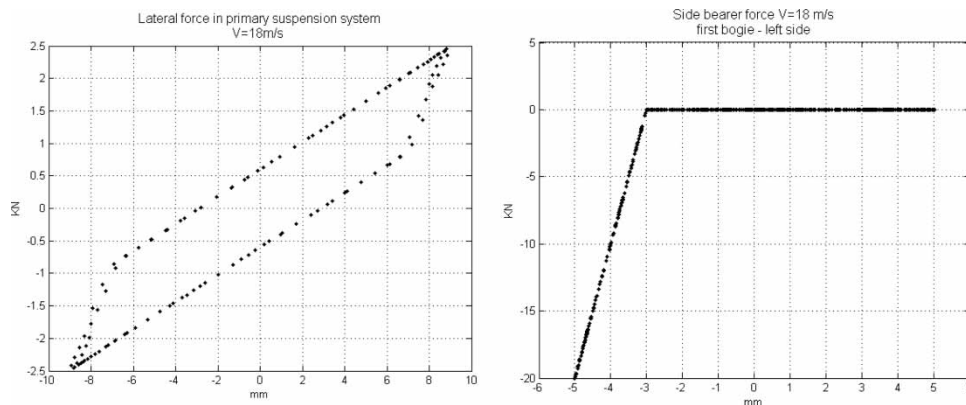


Fig. 11 Lateral force in the primary suspension system and normal force in side bearer (BA652)

To find the critical speed, simulation has been started from speed 25 m/s and after 10 s the speed has been reduced, to 0.02 m/s. Non-linear critical speed for this wagon is 13.8 m/s (50 km/h). As Fig. 9 shows all lateral displacements jump down by 13.8 m/s. The left side of Fig. 9 shows the change in the limit cycle diagram of carbody with speed.

In Fig. 10 positions of two saddle-nodes can be seen. The first one or the critical speed is about 13.8 m/s and the second one is about 15.6 m/s. The speed 15.2 m/s is known as linear critical speed and it is more than non-linear critical speed.

Figure 11 shows lateral force-displacement loops for primary suspension system in the left side of second wheelset at 18 m/s (for 5 s). The damping source is dry friction and the hysteresis loops can be seen in figure. Also Fig. 11 shows normal force in the side bearer. After 3 mm there is solid impact and for solid impacts stiffness of 1.0×10^7 N/m have been used.

Figure 12 shows lateral displacements in limit cycles between 25 and 65 m/s. By increasing the speed, limit cycles change and after about 51 m/s oscillations are chaotic. Figure 13 shows limit cycles in 18 and 64 m/s. In speed 18 m/s the amplitude of lateral oscillation for

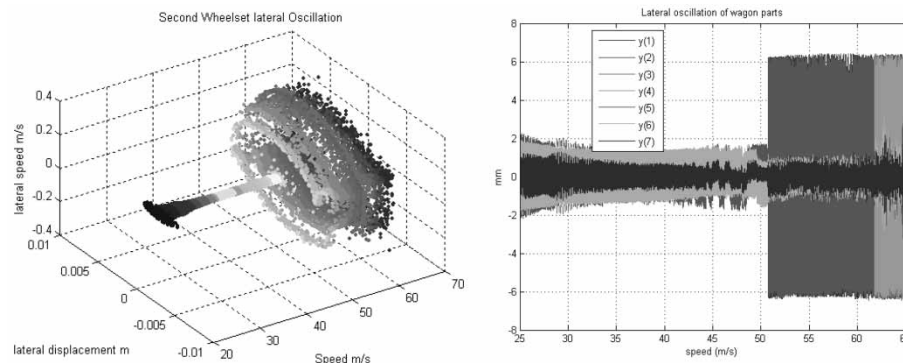


Fig. 12 Changes of lateral displacements and limit cycles by speed (BA652)

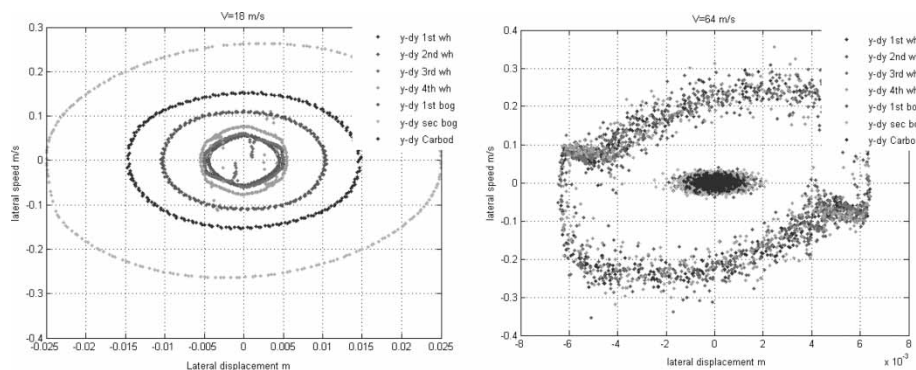


Fig. 13 State-space diagrams for primary and secondary hunting (BA652)

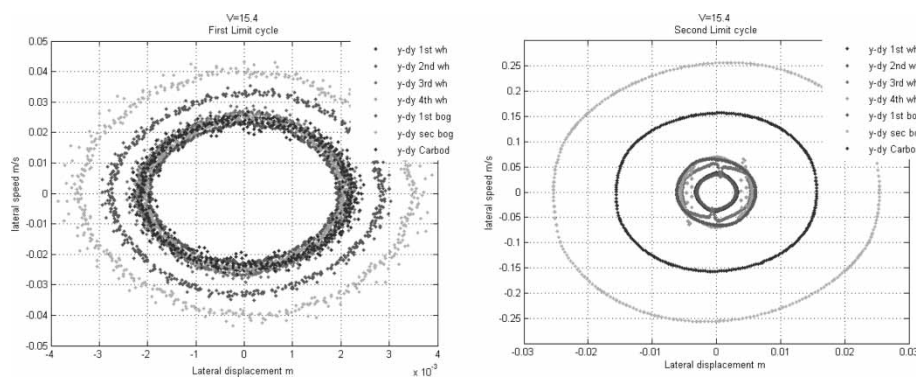


Fig. 14 Two different limit cycles for speed 15.4 m/s (BA652)

bogie frames are more than two times of wheelsets and so wagon is in primary hunting mode. In the 64 m/s, wheelsets have 6 mm lateral oscillations and so they have flange contact. Wagon in the 64 m/s has secondary hunting mode and oscillations of bogies are less than the wheelsets.

In Fig. 14 two limit cycles for 15.4 m/s have been showed. For the second limit cycle, simulation has been started from speed 17 m/s and after 10 s the speed has been reduced to 0.1 m/s to reach 15.4 m/s and in this speed wagon has been simulated for 100 s. As figure shows there are completely different

conditions between these limit cycles. Concentrating in the second limit cycle (oscillation of first wheelset) and comparing with Fig. 10, shows that the responses of wagon depend on initial conditions.

4.2 Critical speed and limit cycles for BA661

Results for BA661 show that there are limit cycles in low speeds and the condition is very critical. For 3 m/s the oscillation of perturbed wagon decreases after 100 s. Figure 15 shows that limit cycle appears in

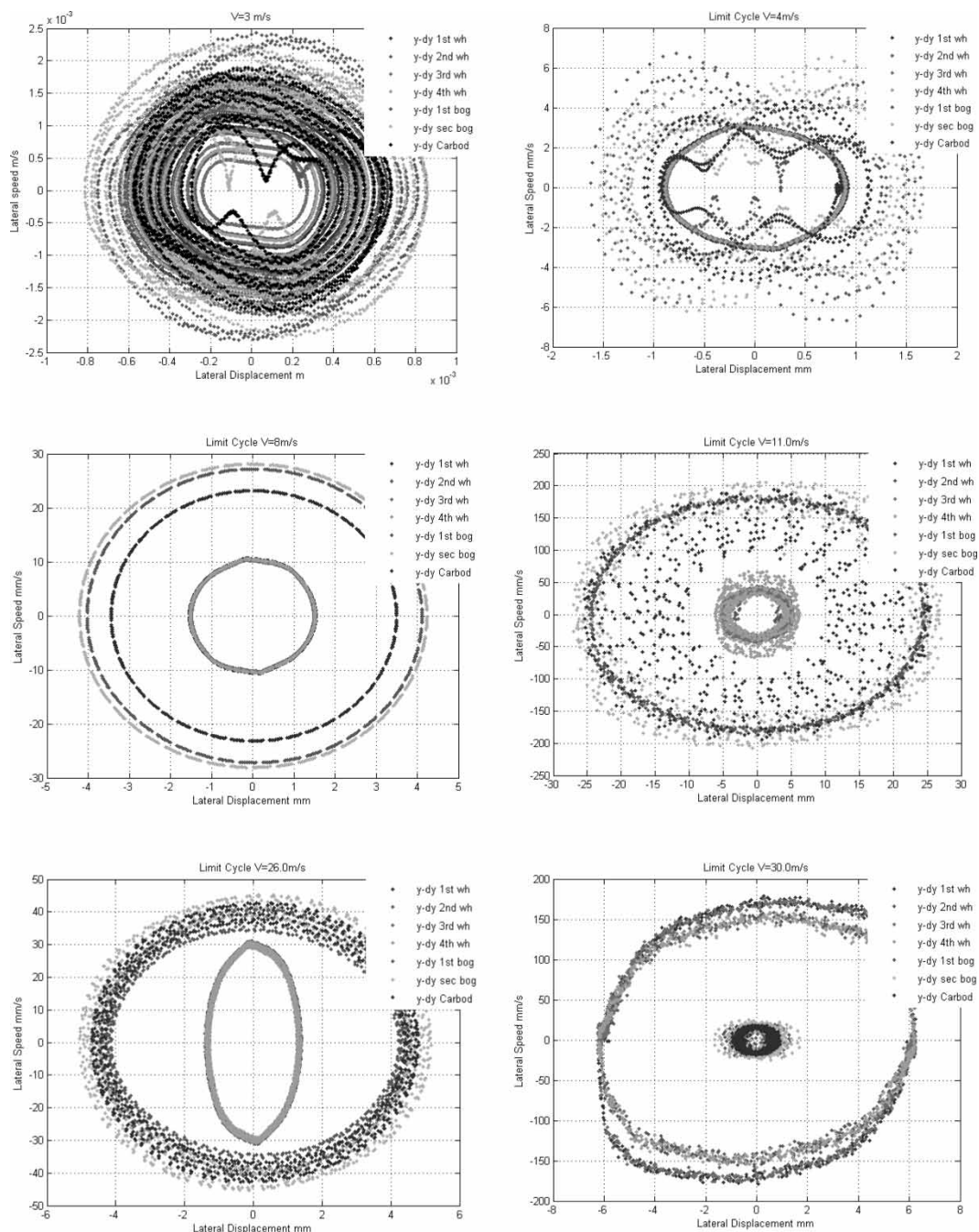


Fig. 15 State-space diagrams for limit cycles in different speed (BA661)

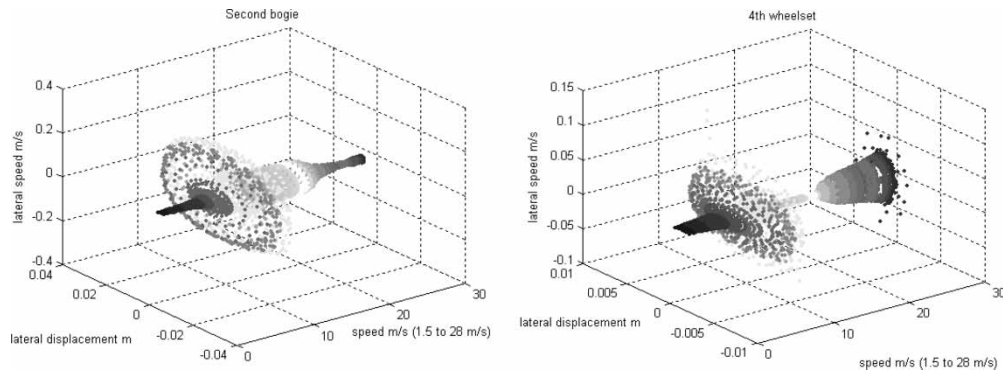


Fig. 16 Non-linear critical speed for BA661 (speed from 28 to 1.5 m/s)

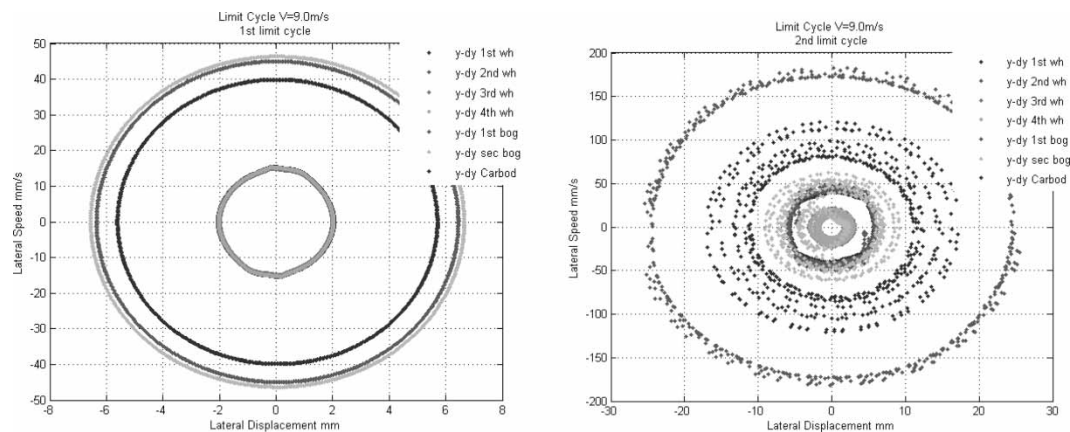


Fig. 17 Two limit cycles for 9.0 m/s (BA661)

4 m/s. In this speed lateral amplitudes for wheelsets are 0.9 mm and for bogies are about 1.5 mm. The amplitude of oscillations increases by increasing the speed to 11.0 m/s. In this speed bogie frames have lateral amplitude of 25.0 mm that is very critical. After 26 m/s lateral amplitudes of bogie frames decrease and lateral amplitudes of wheelsets increase. As have been showed for 30 m/s there are flange contacts between wheels and rails and also lateral amplitude of bogie frames are less than 2 mm.

Figure 16 shows the search for non-linear critical speed. Simulation has been started from speed 28 m/s and after 10 s the speed has been reduced to reach 1.5 m/s. In this figure lateral oscillations of fourth wheelset and its bogie frame are between 28 and 1.5 m/s. As the figure shows there is oscillation in all speeds. Results for different decelerations are the same. Regarding to Figs 15 and 16 the wagon has secondary hunting mode in speeds about 30 m/s and also primary hunting mode in speeds about 11.0 m/s.

Figure 17 shows two limit cycles for 9.0 m/s. The second limit cycle has been calculated by means of decreasing the speed from 12.0 m/s and staying on 9.0 m/s for 100 s.

5 CONCLUSIONS

In the current research critical speed of the BA652-freight truck has been found that is 50.0 km/h. Results show that in dynamical responses of this freight truck there are chaotic limit-cycles and saddle nodes. Also it has been showed that for speeds between 15.2 and 15.6 m/s depend on initial conditions, there are two limit-cycles. Amplitude of oscillations of different parts in limit cycles depends on the vehicle speed. Primary hunting has been happened about 15 m/s and secondary hunting after 51.0 m/s.

For old BA661-freight truck the conditions are critical and carbody can oscillate in very low speeds and with high amplitudes. Both primary and secondary hunting modes can be seen in results and oscillations about 11.0 m/s and after 28.0 m/s are chaotic.

In dynamical models for these two freight bogies, only the characteristics of primary suspension have been different that can be seen in Table 2. Both primary suspension systems have been tested with the same test structure. Comparing results of simulations for these two sets of parameters shows that primary suspension must be checked carefully in the

maintenance centre. Frictions between leaves, bolt and parabolic spring, bolt and chain stone, chain link and chain stone and also the stiffness of leaf spring must be enough to avoid instability in freight wagons.

The same analyses for Y25 freight bogie [10] show that the critical speed is 61.2 km/h. Also in the primary hunting mode the lateral displacement of carbody for Y25 freight truck is less than 10 mm but for BA652 it is more than 15 mm. In the primary suspension, longitudinal clearance for Y25 is 4 mm in one side and for BA652 it is ± 6 mm (12 mm) and also lateral clearance for Y25 is ± 10 mm and for BA652 it is ± 23 mm. On the other hand, tests of primary suspension for these two types of freight bogies show that stiffness in lateral and longitudinal directions of BA652 are less than Y25 (about half of Y25 suspension). Because of these low stiffness and high clearances, BA type bogies are known as steering freight bogies. Results show that increasing the steerability of bogie decreases the hunting speed. In addition to primary suspension, side bearers of Y25 have preloaded springs with frictional pads that help to stability of motions.

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